TRIBOLOGICAL BEHAVIOUR AND TRIBOLOGICAL MODEL OF SHOT-PEENED HELICAL SPRING WIRES

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ABSTRACT

Life and reliability of mechanical units of machines, devices and equipment are determined by a secure functioning of the spring elements. Tribological stresses on coil springs lead to wear and may cause the failure of the spring and the components and thus of the entire unit. The aim of this paper is to clarify the influence of surface roughness on the wear between end coil and change-over coil of the helical compression spring. Tribological tests were carried out with selected surface roughnesses of specifically shot-peened valve spring wires. Various tribological models have been developed to specifically improve the function and life of springs.

Index Terms - friction, wear and tribological model

1. INTRODUCTION

Springs are to be found in very different embodiments in technical products in the Mechanical Engineering, Precision and Electrical Engineering, Automotive Engineering and many other branches [1-11]. Units with helical compression springs and connected components or guidance elements (plate, bushing, pivot, etc.) can also fail when the spring and the surrounding components touch each other during spring compression or extension, resulting in wear [12-20]. With axial spring compression, the coil diameter at block length for fixed and rotatable mounting increases according to Wahl [18].

for fixed mounting $$\Delta D_m \approx 0{,}051\,\frac{S_w^2 - d^2}{D_m} \quad \text{and} \tag{1.1}$$

for rotatable mounting $$\Delta D_m = 0{,}1\,\frac{S_w^2 - 0{,}8 S_w d - 0{,}2 d^2}{D_m} \tag{1.2}$$

where D_m indicates the mean diameter in mm,
ΔD_m the diameter enlargement in mm,
S_w the coil increase in mm and
d the wire diameter in mm.

These occurrences of contact cause mechanical stresses with secondary thermal, tribotechnical, tribophysical and tribochemical reactions between coils and between coils and surrounding components.

In the analysis of the tribological system those contact points within the spring and between spring and spring-surrounding components (Figure 1.1), whose effects on friction and wear are to be investigated were considered particularly.

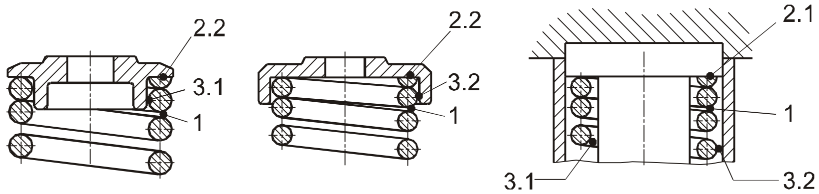


Figure 1.1: Friction and wear points in helical compression springs between: 1 the coils; 2.1 the coil end and the housing; 2.2 the coil end and the spring seat; 3.1 the coil and the hook, eye or loop (where present) or the spring seat; 3.2 the coil and the sleeve or centering pin

Wear leads to changes in the characteristic curve of the spring:

$$R = \frac{G \cdot d^4}{8 \cdot D^3 \cdot n} \tag{1.3}$$

Decreases in wire diameter due to wear lead directly to a change of the rate of the spring. Wear in the area where the end coils are anchored to their surroundings will result in a change to the effective installed length of the spring and thus to a parallel shift in the characteristic curve. These two effects overlap and will result in malfunction if the tolerance limits are exceeded.

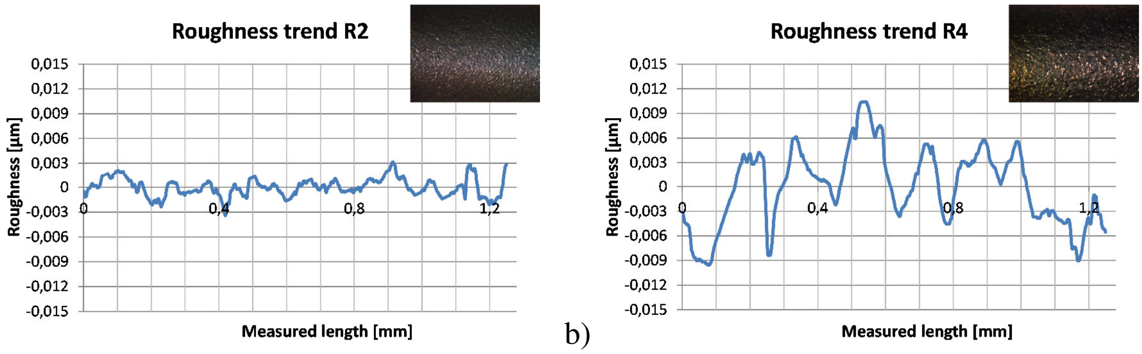
2. EXPERIMENTS TO THE INFLUENCE OF MANUFACTURING "SHOT-PEENED" ON TRIBOLOGICAL BEHAVIOUR OF SPRINGS

Inherent stresses and micro geometry of surfaces play a significant role in friction and wear behaviour. This is particularly true for springs with shot peened surfaces.
To enable the investigation of shot-peened wires a self-developed shot-peening machine for wire was put into operation at the research centre. This allowed the peening of wires, without being crooked. Sample pieces of wire made of oil-hardened SiCr valve spring wire were used for an investigation of the effect of different degrees of roughness on the wear between the end and change-over coils. They were shot peened in an experimental shot peening plant with shot made of special wire. The shot-peening parameters were varied as shown in Table 2.1.
At shot-peening three different pressures were used to generate three different roughnesses. The roughness was measured with a profilometer (Table 2.1 or Figure 2.1).

Table 2.1: Shot-peening parameters used and surface roughness of the wire achieved

Shot diameter [mm]	**Shot hardness**	**Feed [mm/sec]**	**Compressed air [bar]**	**Roughness R_a/R_z [µm] (±0,3 measurement uncertainty)**
0,6	HV 700	1,5	1,3	1,26 / 8,8 (R2)
			2,9	2,54 / 15,2 (R3)
			4,5	3,36 / 21,2 R(4)

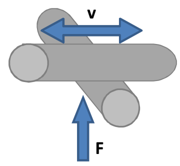
These peened wire pairs with three different surface roughnesses (for example, corresponds to the contact between the end and change-over coil) were tested in experimental model test. To simulate this sort of frictional contact, the experiments were carried out using a special wire mount on a vibration wear tribometer made by Wazau which induces translatory relative movement. Additionally, the operational influences temperature, load, frequency and friction condition (lubricated and non-lubricated) were varied (Table 2.2). The mass difference of the test specimens before and after the test (wear mass) and the mean diameter (linear wear) of the wear surface of the test samples was measured as wear amount.



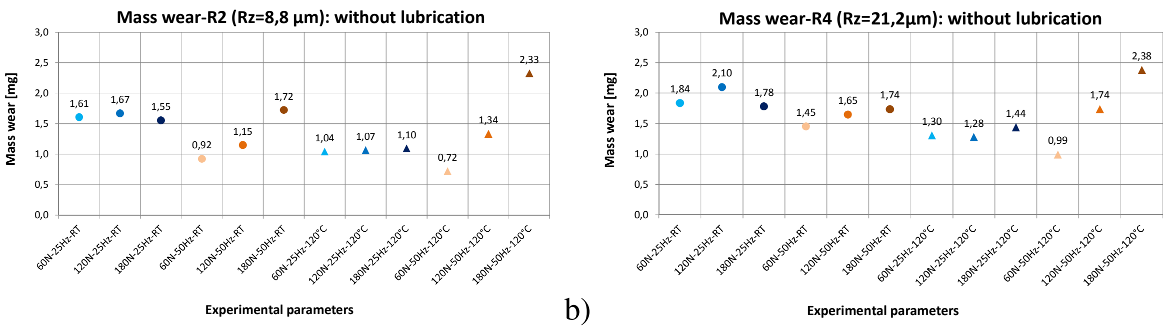
a) b)

Figure 2.1: The example profiles of the surfaces of SiCr valve spring wire:
a) peened with 1.3 bar compressed air (R2);
b) peened with 4.5 bar compressed air (R4)

Table 2.2: Variable operating factors to the experiments

Influential Factor	**experimental level 1**	**experimental level 2**	**experimental level 3**	**experimental model**
Frequency f_0 in Hz	25	37	50	
Temperature T in °C	20	80	120	
Roughness Rz in µm	8,8±0,3	15,2±0,3	21,2±0,3	
Load F_n in N	60±4%	120±4%	180±4%	
Friction Condition	lubricated and non-lubricated			
Deflection: +/-1mm, Experiment Time: 10 min.				

It was necessary to make a realistic friction contact. In the spring load is twisted the wire and enlarged the diameter of the spring. As a result a relative movement between the transition and end coils in the transverse and axial direction of the spring coil is formed. According to the presented experimental plan (Table 2.2) additional investigations at 45 °crossed wires were performed.



a) b)

Figure 2.2: Mean mass wear with a surface roughness of a) Rz=8,8µm and b) Rz=21,2 µm in unlubricated state

Investigations carried out in unlubricated state have shown that with a surface roughness of Rz = 8.8 µm, denoted with R2, at room temperature, at a low frequency of 25 Hz and at the load levels 60N, 120N and 180N, the amounts of wear are almost identical. By changing the frequency from 25 Hz to 50 Hz, the amount of wear depends on the load. By increasing the power the amount of wear increases linearly (Figure 2.2). Furthermore, we found that the friction velocity has a significant influence on the wear. So the mass wear after the experiments at a frequency of 25 Hz was greater than after the experiments at a frequency of 50Hz.

After further experiments at a constant low frequency of 25 Hz, at a load of 60N, 120N and 180N, at room temperature or at a temperature of 120 ° C, the following relationships were found:

A lower wear was found in the experiments with a combination of a smaller load (60N) and a greater frequency (50 Hz) and a higher temperature (120 ° C). A greater wear was found, however, in the combination of a higher load (180N), a greater frequency (50 Hz) and a higher temperature (120 ° C) (Figure 2.2a).

The wear results in the experiments with surface roughness of Rz = 15.2 (R3) and 21.2 µm (R4) show the same dependencies: only the amounts of wear were higher than in the experiments with the surface roughness of Rz = 8.8 µm (R3) with the same factor combinations. That means larger surface roughness resulted in higher amounts of wear (Figure 2.2b).

Comparative experiments were performed on wires with the surface roughness of Rz = 8.8 µm (R2) and Rz = 21.2 µm (R4) in lubricated state. The performed experiments revealed a multiple lower amount of wear.

When determining mass wear in lubricated state the amount of measurement uncertainty was ± 0.02 µg. The measurement errors compared to the small amounts of wear measured were very large. This resulted in the need of determining a linear wear. The linear wear was determined from the smaller diameters d1 and larger diameters d2 in each case, by the mean diameter of the wear domes with an optical measuring device (Figure 2.3d). The measurement uncertainty was about ± 0.017 mm.

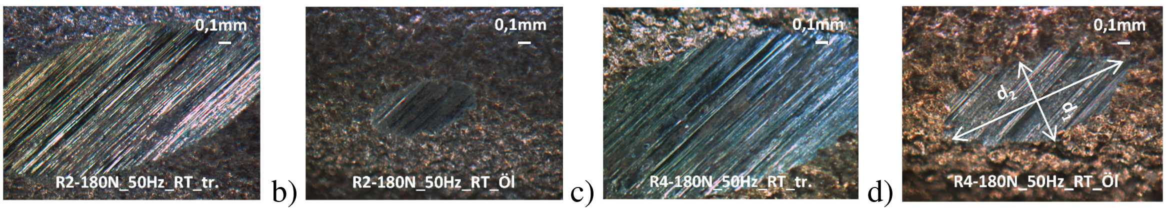
a)


Figure 2.3: Abrasive wear in different experimental parameter combinations

The linear wear changes smoothly in lubricated state with the change of the surface roughness, load, speed and temperature. Figure 2.4a and b show that a combination of a higher load (180N), a greater frequency (50 Hz) and a higher temperature (120 °C) causes a greater wear.

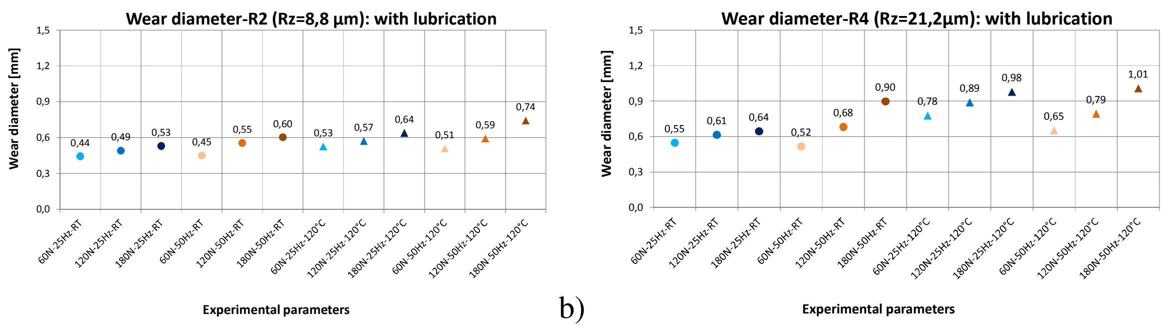


Figure 2.4: Mean wear diameter with a surface roughness of a) Rz=8.8 µm and b) Rz=21.2 µm in lubricated state

Figure 2.5 shows clearly how the coefficient of friction changes at various combinations of influences. Larger surface roughness has higher friction coefficient than smaller surface roughness. The friction coefficient of the experimental combination at room temperature is significantly different from the coefficient of friction of the experimental combination at a temperature of 120 ° C, respectively in lubricated and unlubricated state.

Furthermore, the higher temperatures of 120 ° C in the lubricated state generate a non-uniform profile of the coefficient of friction (Figure 2.5).

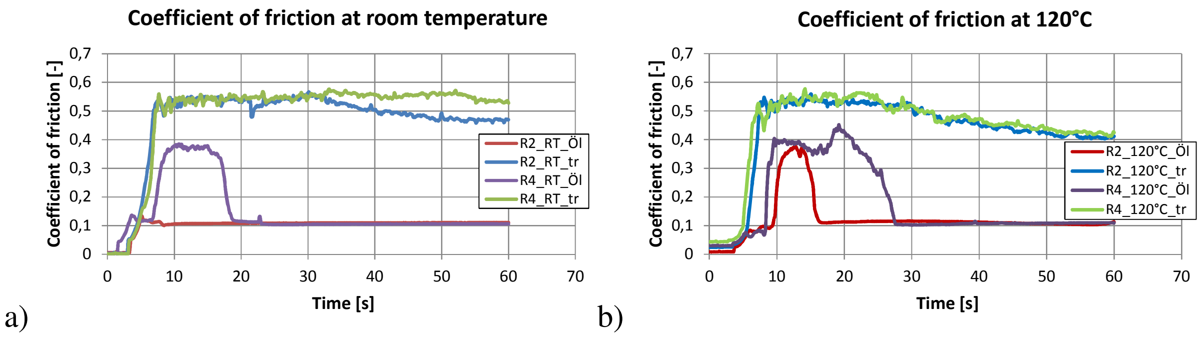


Figure 2.5: Coefficient of friction with surface roughness of Rz = 8.8 microns and Rz = 21.2 microns in lubricated and unlubricated state a) room temperature and b) temperature of 120 ° C

3. TRIBOLOGICAL MODEL OF SHOT-PEENED HELICAL SPRING WIRES

The results of the experimental model tests (tribometer tests) have been systematized and processed with the aim of creating a tribological model. The aim was the creation of tools, which should help the designer to use them for the implementation of the results found.

The investigations described in section 2 have shown that the wear depends on various influencing factors, namely:

the speed of the contact friction,
frequency,
temperature,
the surface roughness,
the material properties,
of lubrication,
the experimental time,
the friction path, etc.

This information has been collected and arranged to determine the most important factors, and finally to describe the wear depending on the input factors mathematically (Y = f (X1, ... Xn)).

It taken into account whether linear or non-linear relationships between factors and target size such as interactions between the influence factors are to be expected [15].

The evaluation of the experiments shows that the wear as a function of influence factors mostly varies almost linearly. So a linear wear model was developed.

The most important prerequisite for creating a mathematical model of wear is the prediction of the investigated influential factors. The resulting experimental design allows to estimate the parameters of a model with the greatest possible precision in a minimal number of experiments and is only valid within the selected test space [16].

Linear wear model

The first step is to select a model in the form of a linear polynomial, which allows the description of linear effects:

$$Y = b_0 + b_1 x_1 + b_2 x_2 + \cdots + b_n x_n \qquad (3.1)$$

by the coefficients b_0, b_1, b_2,…b_n, which are determined from the test results (Wear Y) to the influential factors x_1, x_2,…x_n .

To create the linear wear model some of the experiments (Section 2) to the factors influencing frequency, temperature and surface roughness (Table 3.1) were selected.
With the model from equation 3.1 and the parameters from Table 3.1 an experimental design which includes eight attempts was created.

Table 3.1: Investigated influential factors, load (120N), amplitude (2mm), and run-in period (10 min) constant, non-lubricated condition:

Influential Factor	**Values**
Frequency F_0 in Hz	25 und 50
Temperature T in °C	RT und 120
Roughness Rz in µm	8,8 und 21,2

For the investigated influential parameters two levels for each, that means a fully factorial design at two levels (2k) (k is the number of factors) were determined (Table 3.2).

Table 3.2: Influential parameters to the full factorial experiments at two levels (2k)

Test Number (j)	**Frequency X_1 (F_0 in Hz)**	**Temperature X_2 (T in °C)**	**Roughness X_3 (Rz in µm)**	**Mass Wear Y [g].**
1	25 (-1)	120 (+1)	8,8 (-1)	0,00106
2	50 (+1)	RT (-1)	8,8 (-1)	0,00117
3	25 (-1)	120 (+1)	21,2 (+1)	0,0013
4	50 (+1)	120 (+1)	8,8 (-1)	0,00133
5	50 (+1)	120 (+1)	21,2 (+1)	0,0016
6	50 (+1)	RT (-1)	21,2 (+1)	0,00163
7	25 (-1)	RT (-1)	8,8 (-1)	0,00174
8	25 (-1)	RT (-1)	21,2 (+1)	0,0022
(-1) bzw. (+1) minimum and maximum values of the influential factors.				

The results from the experiments described in section 2 were analysed by means of multiple linear regression. The aim of the multiple regression is to determine the coefficients of equation (3.1) to minimize the sum of the squares of all the residuals “e” for all test results. (Residuals are the difference from the predicted target size and the measured target size.)

$$e = \hat{Y} - Y \tag{3.2}$$

The coefficients of equation (3.1) are represented by:

$$b_0 = \frac{\sum_{i=1}^{N} Y_j}{N}, \qquad b_i = \frac{\sum_{1}^{N} Y_j X_{ij}}{N} \tag{3.3}$$

In this case, N is the number of attempts, Y_j the mass wear of each attempt j and n the number of the influential factor. X_{nj} represents the positive and negative values of the mass wear concerning each influential factor n and attempt j.
Underlyings were selected to express the values of the influencing factors and results in dimensionless units. In this connection X_{1B}= 25Hz, X_{2B}= 100°C, X_{1B}= 15µm such as Y_B=0,001g and the values of each factor are divided by the underlying.
The following coefficients are obtained after the calculation of equation (3.3):
b_0= 1,5; b_1= - 0,07; b_2= - 0,18 and b_3= 0,18.

This linear wear model is only valid for the frequency range of 25 to 50Hz, a temperature range from room temperature to 120 ° C and a surface roughness of 8.8 to 21.2 µm. After comparison of the results a deviation between the experimental and theoretical results of average ± 7 percent was showed.

Square wear model

When investigating an influence quantity at only two levels a possibly non-linear relationship between influencing factors and results in the investigation of can be overlooked. Therefore, additional experiments at middle values, that means at a frequency of 37Hz, temperature of 80°C and a surface roughness of 15.2 µm, were conducted. A quadratic wear model was derived from these and previously determined boundary values. This model describes more accurately the relationships and interactions between the factors due to the possible consideration of possible quadratic dependencies.

$$Y=b_0+b_1X_1+b_2X_2+b_3X_3+b_4X_1^2+b_5X_2^2+b_6X_3^2+b_7X_1X_2+b_8X_1X_3+b_9X_2X_3 \quad (3.4)$$

For the investigated influence parameters each three levels have been established and thus a full factorial design with three levels (3k) (K is the number of factors) has been prepared and carried out.
The calculation of the matrix above yielded the values of the desired coefficients (Table 3.3).

Table 3.3: Calculated coefficients for the calculation of the mass wear with the help of the equation

in unlubricated state			
$b_0 =$ 0,003395333	$b_3 =$ 0,000063109	$b_6 =$ - 0,000000909	$b_8 =$ - 0,000000218
$b_1 =$ - 0,000083531	$b_4 =$ 0,000000712	$b_7 =$ 0,000000395	$b_9 =$ - 0,000000068
$b_2 =$ - 0,000028635	$b_5 =$ 0,000000080		

Substituting these coefficients into the quadratic wear model (equation (3.4)), can determine the mass wear in grams.

$$\begin{aligned}Y=b_0+b_1X_1+b_2X_2+b_3X_3+b_4X_4+b_5X_1^2+b_6X_2^2+b_7X_3^2+b_8X_4^2+b_9X_1X_2+b_{10}X_1X_3\\ +b_{11}X_1X_4+b_{12}X_2X_3+b_{13}X_2X_4+b_{14}X_3X_4\end{aligned} \quad (3.5)$$

X_4 is another factor with a load on the steps of 60N and 180N.

Advanced linear wear model

In the same way an extended linear wear model for lubricated or non-lubricated contacts was formulated (equation (3.5) and Table 3.4).

Table 3.4: Calculated coefficients for the calculation of the linear wear using equation (3.5)

in unlubricated state				
$b_0 =$ 1,990000	$b_3 =$ 0,000000	$b_6 =$-0,000050	$b_9 =$ 0,000120	$b_{12} =$ -0,000018
$b_1 =$ 0,000000	$b_4 =$ -0,003800	$b_7 =$ 0,000330	$b_{10} =$ -0,000130	$b_{13} =$ 0,000021
$b_2 =$ 0,000000	$b_5 =$ -0,000180	$b_8 =$ 0,000002	$b_{11} =$ 0,000090	$b_{14} =$ 0,000011
in lubricated state				
$b_0 =$ 0,460000	$b_3 =$ 0,000000	$b_6 =$ 0,000004	$b_9 =$ -0,000029	$b_{12} =$ 0,000092
$b_1 =$ 0,000000	$b_4 =$ -0,001995	$b_7 =$ 0,000051	$b_{10} =$ -0,000035	$b_{13} =$ 0,000004
$b_2 =$ 0,000000	$b_5 =$ -0,000030	$b_8 =$ 0,000001	$b_{11} =$ 0,000052	$b_{14} =$ 0,000075

Figure 3.1 illustrates the computational and experimental results and confirms by comparison the effectiveness of the previously considered models wear. For a further development of this model more tribological investigations of helical compression springs are needed.

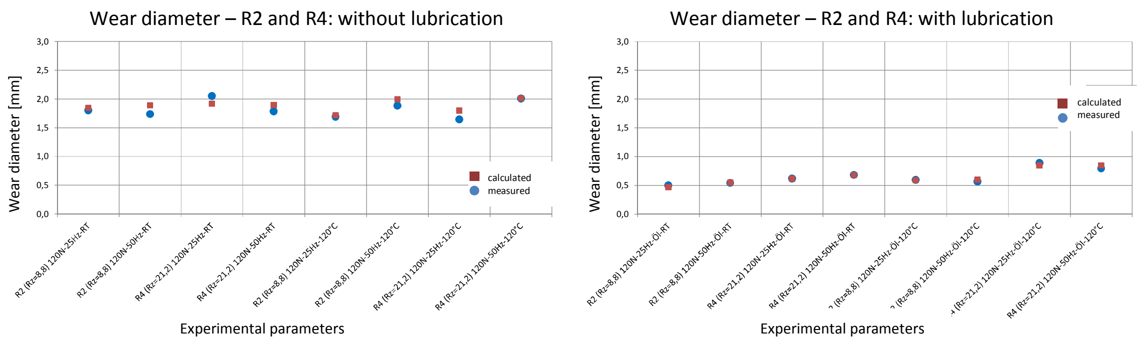


Figure 3.1: Wear diameter calculated and measured a) without lubrication, b) with lubrication

4. SUMMARY

Friction and wear are, as expected, highly dependent on the surface roughness, surface strength and existing residual compressive stresses and heat treatments during the manufacture of the spring (change the wire strength) and surface treatments. The production of springs so has some potential for optimization, there is more research needed.

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